Machinery Messages

Case history

Torsional vibration problems with a synchronous motor

by Peyton Swan, P.E.
Sr. Machinery Diagnostic Services Engineer
Bently Nevada Corporation

A 16,000 HP synchronous motor drove a centrifugal air compressor, in a plant that produced industrial air gasses. The motor ran smoothly in solo tests, but suffered high vibration after it was coupled to the compressor. Between the motor and compressor was a speed increasing gearbox, which increased the motor's speed from 1800 rpm to 5708 rpm. A flexible shim pack-type coupling connected the motor to the gearbox. A solid flange coupling connected the gearbox to the compressor.

During the startup, vibration was high only in a narrow range of speeds (1510 to 1640 rpm), and only occurred as the motor passed through that speed range as it accelerated up to operating speed. At operating speed, the motor ran well. The motor did not have a balance resonance at 1577 rpm; it was not a typical "critical" speed. The vibration itself had an unusual characteristic: the frequency of its dominant component **decreased** as the speed of the motor increased.

The unit was well instrumented (Figure 1). Proximity transducers, mounted at each bearing in an XY configuration, directly observed the compressor rotor and the two gearbox rotors. Motor vibration was sensed with Bently Nevada velocity transducers, because the motor wouldn't accommodate proximity probes. The velocity transducers were also mounted in an XY configuration, at both motor bearing caps. Keyphasor® transducers were installed on both the high and low speed rotors, to provide phase locked speed and phase references. An industry-standard Bently Nevada 3300 Monitoring System processed the transducer signals and displayed vibration levels.The machine train was new, and was ready for commissioning. Its owners contracted with Bently Nevada's Machinery Diagnostic Services (MDS) group, to collect, analyze and document machine data.

Preparation and testing

Prior to our arrival, the motor had been tested while uncoupled from the machine train. Vibration had been low at all speeds, so plant maintenance personnel coupled the motor to the gearbox, and prepared the machine train for testing. We gathered machine data from the outputs on the front of the 3300 Monitoring System. We used an ADRE® 3 System, the predecessor of today's ADRE® for Windows System. ADRE for Windows is Bently Nevada's powerful, portable data acquisition and diagnostic system. It collects and processes up to 16 channels of machine data simultaneously, and displays it on a notebook computer in several different plot formats. ADRE for Windows can process data from a machine at constant speed, (steady state data) and during startups and shutdowns (transient data).

Transient data is extremely valuable in machine commissioning, because it reveals fundamental characteristics of the machine. We use it to assess a machine's operational condition and to identify malfunctions. We include transient data in our documentation, to sup-

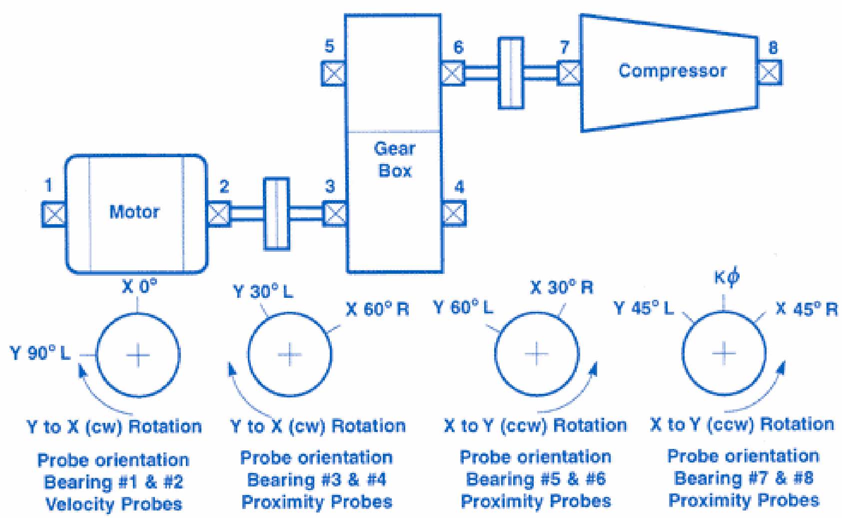

Figure 1
Machine train diagram showing the transducer locations.

port our conclusions and as a baseline record. The baseline helps plant personnel measure changes in machine behavior that could indicate malfunctions, and helps them assess the results of maintenance work.

Operators started the machine train. Vibration was low until the high speed shaft approached 5000 rpm. Then, direct (unfiltered) vibration increased sharply, to over 216 μm (8.5 mils) peak to peak (pp). Over 5000 rpm, vibration decreased, and it reached an acceptable level at operating speed. Maintenance personnel decided that it was safe to perform the compressor load tests scheduled for that day. Operators put the compressor under low load and under 93 percent of full load while we collected steady state data with our ADRE 3 System. When the test was finished, operators shut the machine down while we again collected transient data. Further testing was suspended until we could analyze the data and resolve the problem.

Examining the data

The steady state data we collected at operating speed showed low vibration throughout the machine train. Vibration never exceeded 16 μm (0.63 mils) pp on the low speed shaft, nor 22 μm (0.88 mils) pp on the high speed shaft. Vibration was low at both low compressor load and at 93 percent of full load.

Transient data from the startup, however, did indicate high vibration. Figure 2 is a graph of two different types of vibration data, from the proximity probes mounted on Bearing #6. The solid lines in the top and bottom rectangles are a graph of 1X (synchronous with rotor speed) vibration in Bode format. A Bode plot shows a filtered signal's phase and amplitude; phase in the upper rectangle, and amplitude in the lower. The upper plot in Figure 2 shows little phase information because, at most speeds, the 1X amplitude was lower than the ADRE 3 System's threshold for processing phase information.

The dashed line (Figure 2) is a graph

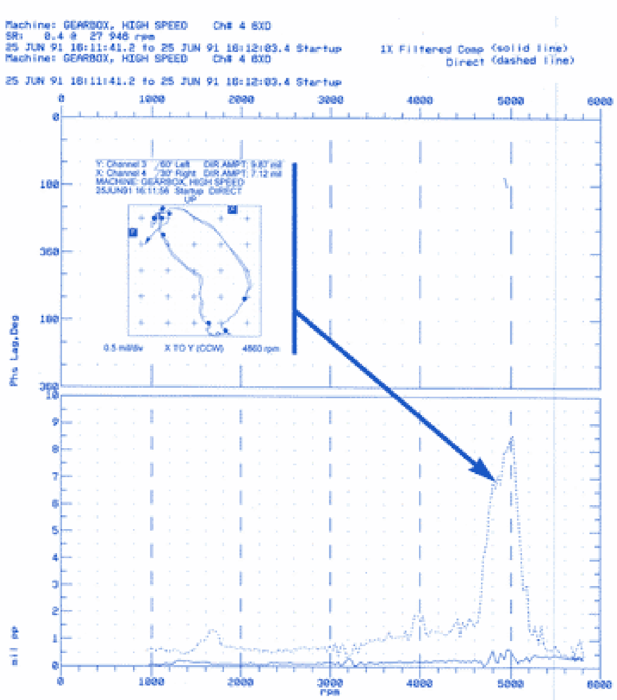

Figure 2
Two types of transient startup data:
(1) The solid lines in the upper and lower rectangles are a Bode plot of 1X vibration. The upper rectangle shows little phase information because, at most speeds, the 1X amplitude was lower than the ADRE 3 System's threshold for processing phase information.

(2) The dashed line in the lower rectangle is an amplitude versus speed plot of direct (unfiltered) vibration. It has no corresponding phase information because it is not a filtered signal.

(3) An orbit plot has been superimposed to show what the orbit would look like at approximately 4900 rpm.

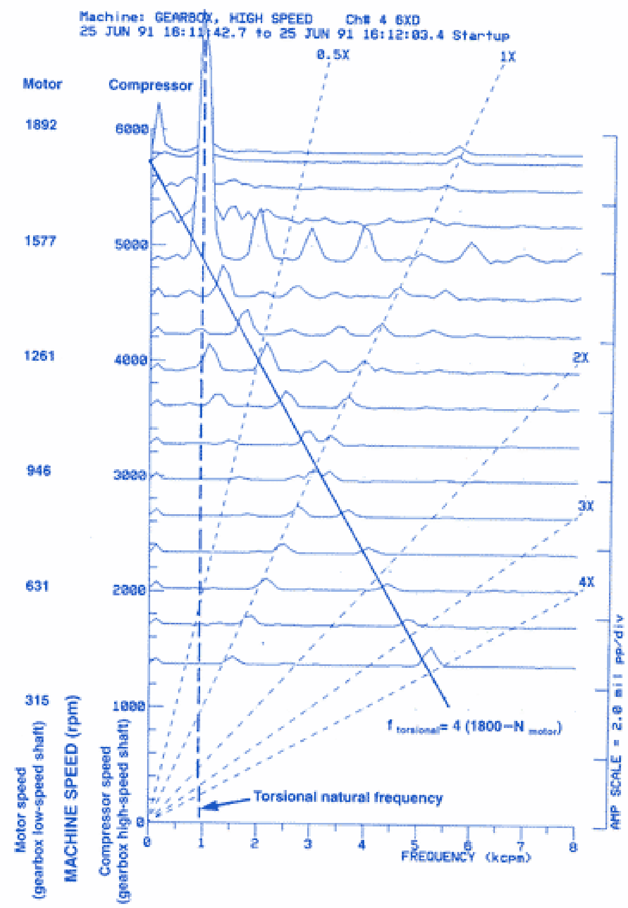

Figure 3
Spectrum cascade plot of vibration on the high-speed shaft, from vibration acquired at startup from proximity transducers mounted at bearing #6. The two vertical scales on the left side are related by a factor of 3.17, the gear ratio of the speed-increasing gearbox.

of direct vibration. It has no corresponding phase information because it is not a filtered signal. The dashed line shows that direct vibration amplitude was low at all speeds, except for a narrow range centered at approximately 4900 rpm. Vibration increased rapidly at that speed, and peaked at approximately 216 μm (8.5 mils) pp. It was as though the rotor had passed through a lateral rotor resonance. However, it was not a 1X balance resonance, because its characteristic, a peak in amplitude accompanied by a 90° phase change, was not present in the 1X Bode plot.

It was clear that the machine's vibration response was at frequencies other than running speed. To identify those frequencies, we examined a spectrum cascade plot of the same data (Figure 3). A spectrum cascade plot is a series of spectrum plots, each of which corresponds to a different shaft speed. Two distinct vibration components are evident. The first is the 1X vibration component. It is typical of the 1X component in most spectrum cascade plots of rotating machinery data. Its positive slope indicates that its frequency increases with the speed of the motor.

The second vibration component is not typical. It has a negative slope, which indicates that its frequency decreases as the rotor speed increases. If the line were extrapolated, the component would have a frequency of 7200 cpm when the motor was first started up. As the motor increases speed, the component's frequency decreases, in inverse proportion to motor speed. The component's amplitude peaks when the motor speed is just above 1500 rpm (and the high-speed rotor reaches 4900 rpm). At that point, its frequency is just above 1500 cpm. Its frequency continues to decrease, until it is 0 cpm when the motor speed reaches 1800 rpm.

That same vibration component was evident in transient startup data from the bearings on the low speed gearbox shaft, although the amplitude was lower. The amplitude was even lower in the similar components contained in signals from the velocity transducers mounted on the motor bearing caps.

We viewed transient startup data in other plot formats, including orbit and shaft centerline. Those plots appeared normal. The shutdown data contained no evidence of the unusual vibration component.

Analysis

Synchronous motors have a characteristic torsional vibration response. The twisting force comes from the rotating electrical field that propels the rotating shaft. It has two components: a steady torque, and an oscillating torque. The oscillating torque is proportional to the difference in speed between the rotating shaft and the rotating electrical field that drives it. When the motor is at synchronous speed, the oscillating torque is negligible. During startup, its characteristic torsional response is described by the following equation:

$$\text{Torque}_{\text{total}} = \text{Torque}_{\text{steady}} + \text{Torque}_{\text{dynamic}} \cos\{(2\pi / 60)P\,(N_{syn} - N_{motor})\,t\} \qquad \textbf{(Eqn 1)}$$

where:

P = number of poles

N_{syn} = synchronous speed

N_{motor} = instantaneous motor speed at any time, t

If we substitute values for a 4-pole, 1800 rpm synchronous motor, the frequency of the torsional vibration reduces to:

$$f_{\text{torsional}} = 4\,(1800 - N_{\text{motor}})$$

This equation defines a torsional vibration with a frequency that decreases as motor speed increases. At operating speed, the oscillating component disappears. It also explains the frequency of the unusual vibration component in the cascade plot (Figure 3).

Although oscillating torque during startup is a characteristic of synchronous motors, it usually doesn't cause excessive radial vibration. Under most circumstances, a motor's torsional vibration cannot be detected with transducers that are configured to detect radial motion. In this case, one of the torque frequencies generated by the motor, as it came up to operating speed, matched a torsional resonance of the gearbox high-speed shaft. The resonance amplified and cross-coupled the motor torsional vibration, causing high radial vibration.

The rotating field that turns the shaft disappears when power is removed from the motor field. Without field current, the motor has no rotating motor field, and lacks a necessary condition for torsional excitation. That is why torsional vibration was not evident in this machine's shutdown data.

Conclusions

The high amplitude radial vibration component on the high-speed gearbox shaft originated as torsional vibration on the motor shaft. Its amplitude increased when its frequency matched a torsional resonance on the high speed shaft. The torsional vibration was cross-coupled on the high-speed shaft, and generated the radial vibration that we measured with proximity probes.

Strong torsional vibration forces can impose severe cyclic stresses on gearing systems, which can lead to early gear tooth failure from high cycle fatigue mechanisms. The high-amplitude radial vibration from a cross-coupled torsional vibration can also cause premature bearing and gear tooth wear. If you have noticed premature bearing and gear tooth wear on machinery, you should explore ways to reduce high radial vibration amplitudes.■

References:

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